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ACV CUSHION COMPARISON TESTS. PRELIMINARY REVIEW AND DEFINITION--ETC(U)
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ACV CUSHION COMPARISON TESTS
Preliminary Review and Definition Of Model and Tests

APRIL 1979

FINAL REPORT

PREPARED FOR:
OFFICE OF NAVAL RESEARCH

Aerojet
Liquid Rocket
Company

Report Number LCAC-2299-002

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ACV CUSHION COMPARISON TESTS, PRELIMINARY REVIEW AND DEFINITION OF MODEL AND TESTS,

9 FINAL REPORT,

11 APRIL 1979

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1. INTRODUCTION

The US Navy/Marine Corp's amphibious assault mission represents an ideal application for the unique characteristics of the air cushion vehicle (ACV).

The purpose of this Navy technology program, directed by the AALC Program Office of the David W. Taylor Naval Ship Research and Development Center (DTNSRDC), is to be able to compare cushion systems.

During the last two decades ACV's have been designed and built to suit many different applications. The advent of skirts or seals, that allowed the cushion height to be increased considerably without proportionate increases in lift power, transformed the hovercraft principle into a viable means of providing air cushion support.

Five major forms of full scale cushion systems have been developed by independent companies. These are the Aerojet-General loop/pericell, the Bell-Aerospace Textron bag/finger, the Bertin multicell jupe, the British Hovercraft Corporation tapered skirt and the Hovercraft Development Limited loop/segment.

In each of the above cases, the general desire has been to produce a cushion system that would provide a craft with ability to operate at relatively high speed over an uneven surface. The constraints have been very different as well as the modes of development, but the end products have a great deal of commonality. However, there is no common data base to allow direct comparison or to select the best configuration for a particular application.

For the amphibious assault application, the size of the ACV is limited by the need to operate from the well-decks of existing amphibious assault lift ships. The payloads are comprised of combinations of already defined vehicles, pallets and other cargo, some of which are very dense and others which require a lot of deck space with clear drive-through capability. It is desired to be able to operate in a range of conditions of temperature, wind, sea state, surf, beaches and terrain.

1. INTRODUCTION (Cont'd)

The "Best" design will be dependent upon not only the performance of the craft under all these conditions but on the total or life cycle costs which includes such things as maintenance and replacement costs as well as the acquisition and direct operating costs.

The prime objective of this cushion comparison program is to be able to obtain data on the various existing cushion systems, that can be compared directly in the context of the amphibious assault mission. Full scale data is very limited and does not provide a good base for comparison. The use of a series of scale models, built and tested to a common standard, will provide a means of obtaining basic performance data that can be compared directly. p. 3

The requirements for such a set of models is addressed in Section 2. The use of the models and the test program is described in Section 3. It was necessary to prepare a preliminary design and description of the model, as reported in Section 4, to obtain an estimated cost and schedule for the model. This is reported in Section 5.

The results of this planning stage of the program will be used to establish the detailed requirements for the design and building of the models, the test program and the comparison of the cushion systems including full scale considerations. The cost and schedule guidelines will also be established.

The draft of this report was submitted and compared with the corresponding draft reports for the other model types. As a result of the comparison, a coordinated set of model design criteria was prepared by the Navy Technical Office (Ref. 9070 1182:2GW, dated 26 January 1979) and forwarded for consideration. This report now includes changes that reflect consideration of the given design criteria.

2. MODEL REQUIREMENTS

The model is to be a dynamically similar representation of an air cushion vehicle (ACV) designed to the requirements of the Navy's amphibious assault mission. The purpose of the model is to provide data for the comparison of cushion systems.

A review of the test requirements in Section 3, in conjunction with the requirements for the model, resulted in the selection of a linear scale factor of 1/12 full scale for the model was selected.

The features considered most important are the flexible skirt that contains the cushion, the lift fan system and the lift air distribution system together with the general hull form.

The amphibious assault lift ships, that are to carry the ACV's, impose strict limits on the major dimensions of the craft. These are to be reflected in the model. These dimensions are given in Table 1, which summarizes the model requirements and criteria, and are illustrated in Figure 1.

The skirt is obviously of prime importance as the object of the program is to make a comparison of different cushion systems. It is recognized that proper representation of all the structural characteristics of the skirt is not possible within the context of Froude scaling. This is primarily due to the fact that the full scale materials have low valued elastic constants and are subject to severe distortions under static and dynamic load conditions. The selection of materials for models is limited and the model skirts must generally be constructed of woven nylon fabrics of the same physical properties as the full scale. For dynamic similarity the properties should be appropriately scaled. Young's Modulus, for instance, should vary as the linear scale factor. The density is correctly the same regardless of scale. The thicknesses should be scaled and then the weights would be in the right proportion.

For inflated membranes, the tensions are related to the local curvature and pressure difference across the surface. Some errors in shape and action result from non-scaled elongations due to these membrane tensions.

TABLE 1

Summary of Model Requirements and Criteria

Requirement

1. GENERAL

The model should be a dynamically similar representation of an ACV with the following characteristics:

	Full Scale		1/12 Scale	
a) Overall length, with cushion inflated	29.2608 m	(96 ft)	2.4384 m	(96 in.)
hard structure	27.8777 m	(91.4583 ft)	2.3231 m	(91.46 in)
b) Overall beam, with cushion inflated	14.6304 m	(48 ft)	1.2192 m	(48 in.)
hard structure	13.1064 m	(43 ft)	1.0922 m	(43 in.)
c) Payload deck width	7.9248 m	(26 ft)	0.6604 m	(26 in.)
d) Hull depth, at main deck	1.5240 m	(5 ft)	0.1270 m	(5 in.)
at superstructure	2.1590 m	(85 in.)	0.1799 m	(7.08 in.)

2. SKIRT

It is recognized that proper representation of all structural characteristics is not possible within the context of Froude scaling. However, the skirt shall represent the significant design features of the full scale skirt including:

a) Geometry of all component panels and joints

b) Attachments

c) General material type

3. LIFT FANS

The lift fans and housings must fit within the hard structure and be outside of the payload deck, a width of

2.7432 m (9 ft) 0.2286 m (9 in.)

The lift fans must provide appropriate pressure and sufficient flow for optimum operation under all conditions. Since this optimum will be determined during the course of the model tests a range of ± 25 percent around a nominal flow shall be provided

Flow required,

Minimum	283 m^3/s	(10,000 cfs)	0.567 m^3/s	(20 cfs) s)
Maximum	453 m^3/s	(16,000 cfs)	0.908 m^3/s	(32 cfs)

The lift fans shall be powered by 3 phase, nominal 400 Hz variable frequency motor(s).

Note: The use of the four decimal places in the metric values was to allow an exact conversion from English units and does not imply a particular accuracy in the measurement.

Criteria

The linear scale factor shall be 1/12.

The features considered most important are the flexible skirt that contains the cushion the lift fan system and the lift air distribution system together with the general hull form.

All general dimensions of the model hull and skirt shall be within a tolerance of ± 1.5 mm (±1/8 in.)

Critical interfaces such as fan to duct, or fan machinery shall be more accurate as required.

It is presumed that all full scale skirts are constructed from woven fabric of nylon with an elastomeric coating to provide air tightness and wear resistance.

For convenience a common material is suggested for all models, namely "Belflex 40". The material shall be procurred by one contractor for the use of all. Similarly, the bonding process and adhesives shall be common to all models. A yellow, orange or red color is to be preferred to give the maximum visibility in the presence of water and spray.

Fan machinery shall be geometrically similar to that required for the full scale craft.

Scaling imperfections shall be compensated for by use of increased fan operating speed.

Model air flow and pressures will be determined by appropriate calibration of the complete system.

TABLE 1 (Cont'd)

Summary of Model Requirements and Criteria

Requirement

4. **HULL**

The model shall be capable of representing a range of full scale load conditions including weight, cg location and moments-of-inertia.

a) Weight.

	Full Scale	1/12 Scale
Minimum	131 500 kg (290,000 lb)	76 kg (168 lb)
Maximum	176 900 kg (390,000 lb)	102 kg (226 lb)
Bare model (See criteria)		57 kg (126 lb)

b) Center-of-gravity

Adjustable ±3 percent of model length and beam.

c) Moment-of-Inertia at nominal weight of 154 200 kg (340,000 lb) 89 kg (197 lb)

Pitch	9.26×10^6 kg-m^2	(6.83×10^6 slug-ft^2)	37.22 kg-m^2	(27.45 slug-ft^2)
Roll	2.66×10^6 kg-m^2	(1.96×10^6 slug-ft^2)	10.67 kg-m^2	(7.87 slug-ft^2)
Yaw	11.06×10^6 kg-m^2	(8.16×10^6 slug-ft^2)	44.46 kg-m^2	(32.79 slug-ft^2)

Criteria

The total model weight includes 9 kg (20 lb) allowance for towing gear (heave pole) and instrumentation and 10 kg (22 lb) allowance for ballast.

The weight shall be within ±0.1 kg (±0.2 lb)

The location of the ballast shall allow an accuracy in moment of ±0.01 m-kg (±1 in.-lb)

The nominal location of the center-of-gravity shall be at deck level above the center-of-area of the projected cushion in the ground plane. This point shall be used as a reference point for measurement of locations, forces moments and moments-of-inertia for all tests.

The location of the ballast shall allow an accuracy in moment-of-inertia of ±1 kg-m^2 (±0.7 slug-ft^2)

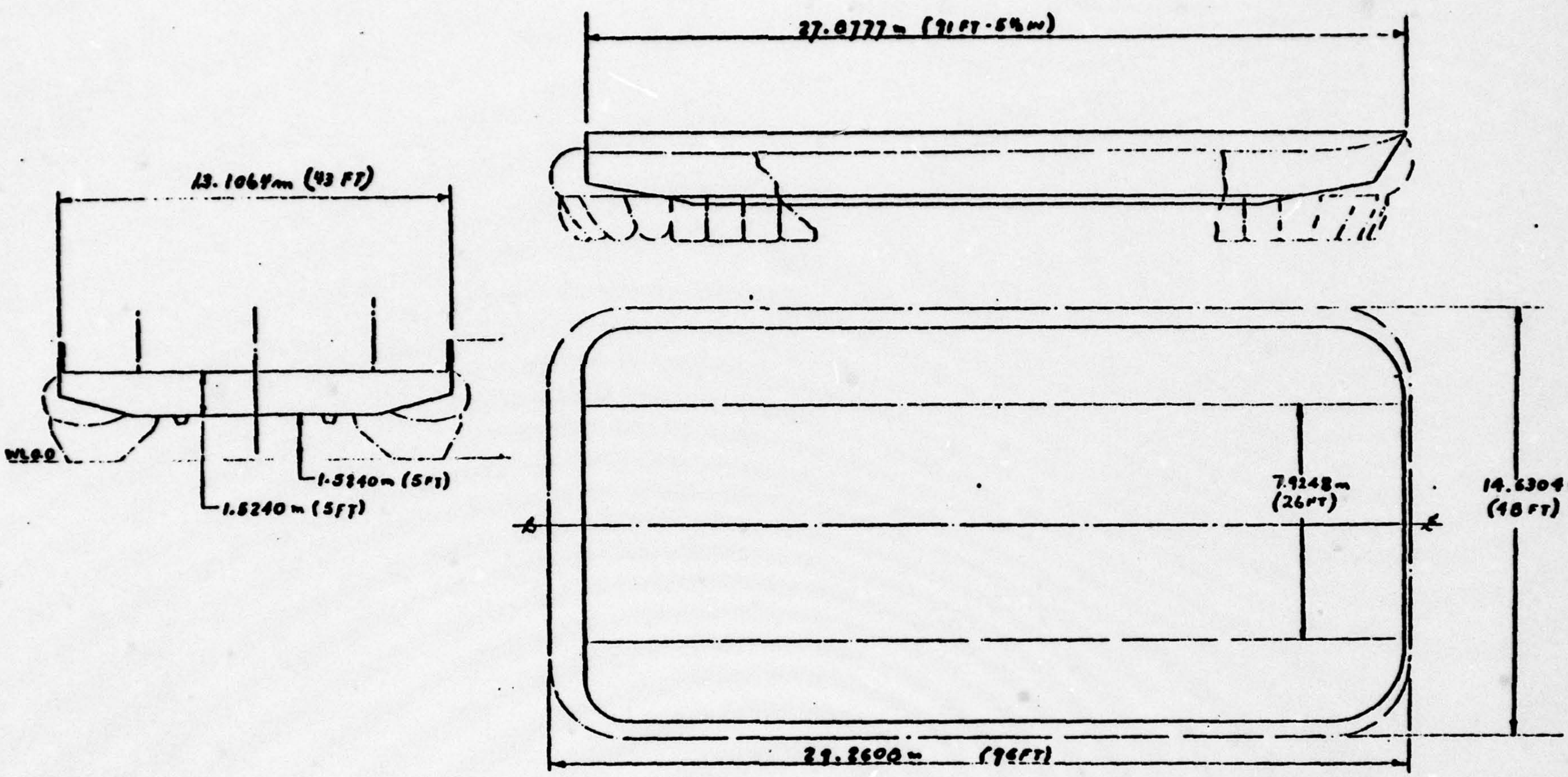

Figure 1 Base Line Full Scale Hull Dimensions For Cushion Comparison Model

2. MODEL REQUIREMENTS (Cont'd)

For members that have significant thickness, such as doublers and stiffeners, bending loads can be appreciable and are dependant upon the Young's Modulus, which as discussed above will not be to scale. This results in model skirts being much stiffer than is appropriate to full scale. The errors may be minimized by using the thinnest possible material and omitting doublers and stiffeners where appropriate. Bonded joints are usually much wider than scale, because of limitations in fabrication technique and the adhesives that add further stiffness.

These scaling problems are probably the greatest shortcoming in attempting to compare skirts through model tests as the errors are virtually impossible to be compensated for on any theoretical basis. Empirical correction factors may only apply at certain scales for certain configurations under special conditions and are not universally available.

Since all skirts rely on the lift air pressures for inflation, and this will be similar for models of the same scale, under the same conditions, it would seem appropriate to use the same material for all skirts, to minimize the difference between the different models. The suggested material is "Belflex 40"; a lightweight, 70 denier nylon fabric of approximately 1.7 oz/sq yd proofed to a total weight of 2.3 to 2.4 oz/sq yd. This compares to the 18 oz/sq yd nylon fabric used on both JEFF(A) and JEFF(B) that was coated to a total weight of 50 to 90 oz/sq yd. It is evident that this material has greater than scale stiffness but is considered to be of the minimum workable weight.

The lift fan system is extremely important as this supplies the air to the cushion and contributes to many of its properties. The maximum lift capability is obviously determined by the pressure that is provided by the fan. Similarly, the ability to traverse an uneven surface such as waves is dependent upon the ability to replace lost cushion air with the air flow generated by the lift system.

The overall power required to operate the craft under given conditions, is dependent upon the efficiency with which the necessary pressure and flow can be provided and also the effect that the cushion has on reducing drag and hence the propulsive power demands.

2. MODEL REQUIREMENTS (Cont'd)

The cushion must also provide appropriate stability characteristics to react the effects of wind and operation over waves or uneven surfaces, the craft generated propulsion and control forces and non-optimum location of cargo loads. The dynamic characteristics of the cushion system determine the craft motions that will occur in response to operation over uneven surfaces. These cushion static and dynamic characteristics are affected by the lift fan and the distribution system including the air distribution within the skirt. The design operating point on the fan head flow characteristic curve, the slope at this point,and the shut-off pressure or pressure at zero flow are generally accepted to be the key characteristics. The performance of scaled fans of very small size (3-4 inches in diameter) is inevitably less than that of their full scale equivalent. The losses in the model ducts is usually higher so an increase in fan operating speed is recommended to compensate for the losses. Calibration of the total system is important to establish the actual performance obtained so that full scale equivalent conditions can be determined.

The hull is of less significance to the cushion comparison tests than the skirt or lift system as normally there is no contact between the surface and the hull. However, the geometry of the skirt is dependent upon the hull shape and the inflated geometry of the full scale craft is restricted by the maximum height, length and beam limitations of the lift ship well decks. In extreme conditions the hull may contact the surface and therefore its shape in potential impact areas is of concern. The shape depicted in Figure 1 shows inclined plating on all edges of the hull bottom, and at the bow. The angles must be considered in conjunction with the form of each skirt and so cannot be specified. Similarly, some form of landing rail or pads should be included. A relatively large landing pad area is required because of the limited load bearing capabilities of the ship well deck, and of unprepared landing sites.

The radii at the hull corners can be varied to suit the skirt,but it is desirable to leave the maximum possible space within the hull for machinery andother items. Also, it will be more convenient to have straight ramps across the full payload deck width.

The bow ramp shall be 3.048 m (10 feet) above the baseline and the stern ramp 2.4384 m (8 feet) above the baseline. The sidewall height shall also be at least 3.048 (10 feet) but a maximum of 3.81 m (12.5 feet)

The 1.524 m (5 foot) hull depth, from hull bottom to payload deck is that used by both JEFF(A) and JEFF(B) and was arrived at in consideration of maintaining overhead clearances and yet minimizing structural weight. For the model it is essential to minimize weight changes due to water collecting in voids on other water traps, therefore, the model should be completely foam filled.

In general there is little problem in constructing scale models of adequate overall strength. There is need to consider how to take the local loads due to the skirt attachments, lift fan machinery, propulsor(s) (if used) instrumentation, ballast and the various test rigs. Previous Aerojet models of this type have included continuous major full depth longitudinal structural members, spaced 0.0889 m (3.5 inches) either side of the model centerline. Additional longitudinal, transverse and edge members are provided for the other local loads. This arrangement would be compatible with the 0.1524 m x 0.1524 m (6-inch by 6-inch) and 0.1524 m x 0.2032 (6-inch by 8-inch) pad requirements for tow gear that were called out.

Because of the magnitude of the loads, it is suggested that 0.008 m (1/4-inch) Aluminum plates be used in place of the 3/4 inch mahogany specified. These plates can be very securely mounted to the model by means of through bolts in the major longitudinal members. They can also be removed and replaced for convenience in making changes and drilling and tapping holes to suit particular fittings.

Experience has shown that is best to tow models through a fixed reference point close to the model center-of-gravity. The added realism obtained by towing through a higher point representing the centerline of the propulsors is offset by the adverse pitching (and/or rolling) moments that result from using higher than full scale equivalent forces to accelerate the model.

2. MODEL REQUIREMENTS (Cont'd)

A fixed reference point is more convenient than attempting to relocate the tow gear for every change in model ballast condition. For further convenience the center section of previous models have been left without foam filling to allow clearance for pitch and roll gimbals and force measuring block gages.

The outer skin of models have been constructed of plywood on occasion but fiberglass has been found to be more durable and is preferred. The internal members can be of suitable wood, including plywood, as they can be well protected. Pressure ports and other instrumentation can be secured by means of screws and/or epoxy resin or similar methods as required.

3. TEST PROGRAM

The selection of the "Best" cushion system involves the consideration of many factors. The comparison of a single performance characteristic of the different systems, such as model drag under a particular set of external operating conditions, appears simple enough. It soon becomes apparent that even this is not really simple because of the number of parameters involved in even setting the model operating conditions. The most logical basis for comparison would be the full scale design condition. This might be expressed in terms of operation at a given speed in a certain wind and sea state condition. The scaling and controlling of the forward speed is straight forward. The wind cannot be represented in the tow tank and even if the full scale sea state can be defined there are problems in representing it. For instance, only long crested waves can be generated in the linear tow tank. These are thought to be more adverse than a real sea with an equivalent wave energy point spectrum.

As far as the model conditions are concerned, each should be run at it's own optimum condition which includes selection of lift system setting and trim or center-of-gravity location. To establish this will take a series of runs for each model.

Perhaps the most meaningful measure of comparative performance becomes the total power required for both lift and propulsion. This requires knowledge of propulsive efficiency to convert a measured drag to required thrust and hence power. Also the lift system efficiency is required to convert lift air flow setting to an equivalent power. If all these details were known then one could make a reasonable judgment for that set of conditions.

The next problem is how to deal with other basic performance characteristics such as ride, controlability and sensitivity to off-design conditions. An example here is that one cushion system may have a very low total power requirement if say the center-of-gravity is adjusted within fine limits but then require a considerable increase in power if the center-of-gravity is moved. What if the alternative system that is being compared requires

a little more power than the first system at its optimum condition but does not suffer as large an increase in power requirement for the same specified center-of-gravity shift resulting in lower total power requirements at the extremities of the range. Maybe the average drag over a required range would be the most appropriate measure. If a mission profile were defined it may be more appropriate still to weight the significance of some part of the total range of parameter variation. All this may be rather academic if the performances are radically different. Until at least preliminary test results have been obtained, the decision to run more detailed tests should not be made.

Having conducted these basic performance measurements at or around design conditions, it will be important to determine that other important requirements can be met. Ability to accelerate through hump without excessive increase in power, provide an acceptable ride, and be readily controllable without fear of divergent motions such as plow-in, broaching in surf or yaw build-up during turns leading to pirouetting or even overturn. These phenomena are very difficult to evaluate directly from model tests as they depend upon the craft control characteristics and having an operator in the loop.

Even if all desired tests can be run on the models, and suitable methods are available for the interpretation of the results there still remain questions of how to convert to full scale and how to evaluate other aspects such as wear, accessability for maintenance, failure modes, acquisition costs and life cycle costs. The total impact of the cushion system selection on the ability to fulfill the mission is even more difficult to assess as it includes such considerations as vulnerability and availability. Full scale data will be essential before all of these effects can be taken in account.

The suggested test program has been reviewed on the basis of the discussion above and the various options and comments are summarized in Table 2.

4. MODEL DESCRIPTION

The requirements for the model have been described in Section 2 above. These have been based on considerations of model scale and of the test program.

The major dimensions and other key parameters of the full scale craft are given in Table 3 together with the corresponding model values based on linear scale factors of 1/10, 1/12 and 1/14. Generally the larger the size of the model, the smaller the errors due to imperfections introduced by scaling. To be compatible with the tow tanks and the other facilities at DTNSRDC at Carderock, Maryland, a 1/10 scale probably represents the largest practical model that should be considered. Models of the Amphibious Assault Landing Craft (AALC) JEFF(A) and JEFF(B) have already been tested at 7/100 scale (1/14.3 scale) and 1/12 scale, respectively. The smaller scale of the JEFF(A) model was selected to be compatible with the tow tank facilities of the Stevens Institute of Technology in Hoboken, New Jersey. The larger scale of 1/12 was selected for the JEFF(B) model and most of its tests were conducted at Carderock. This scale of 1/12 appears to be the best compromise for the tests envisioned and to be compatible with the DTNSRDC facilities.

From the requirements described in Section 2 and the general geometry of Figure 1, a more detailed layout has been prepared, Figures 2a and 2b include structural details. The construction concept employed has proved to be very effective on a number of similar models. A simple plug is made which represents the hull form. A plaster mold is then made from the plug. The outer skin of the model is then formed inside the mold from fiberglas in a manner similar to that used for the majority of modern small boats. This outer skin provides a watertight shell for the model. An alternative method is to construct a female mold directly.

The internal structure is bonded to the skin. It is configured to take the model towing forces and to support the lift and propulsion machinery and to provide overall bending and torsional stiffness. Members are added to take local loads such as the skirt attachments. This internal structure is most conveniently fabricated from wooden strips with plywood webs as stiffeners. Some parts, such as air ducts, can more conveniently be made from light gauge sheet metal or fiberglas.

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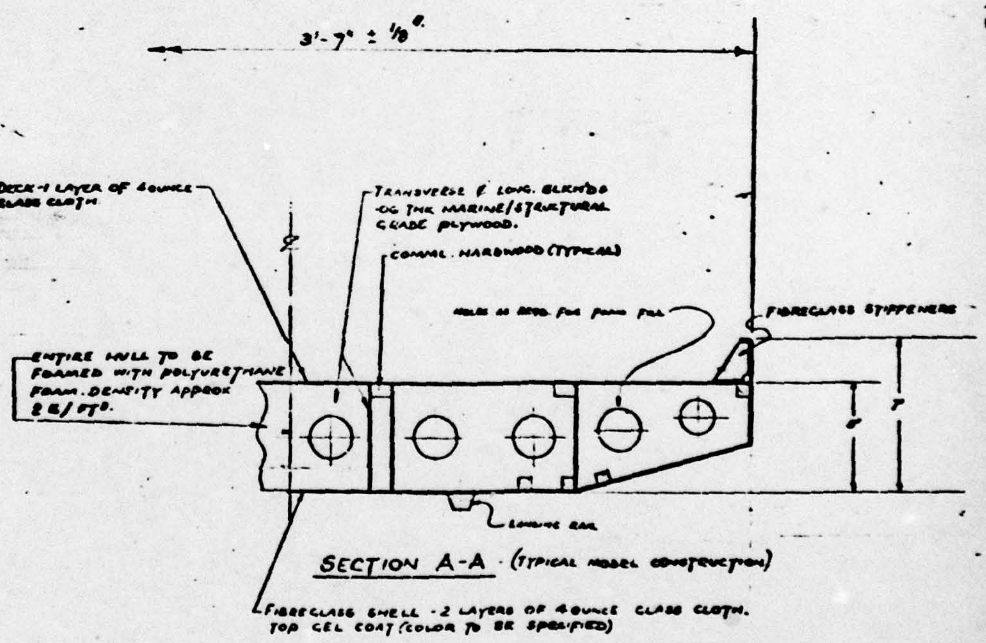


SECTION A-A (TYPICAL MODEL CONSTRUCTION)

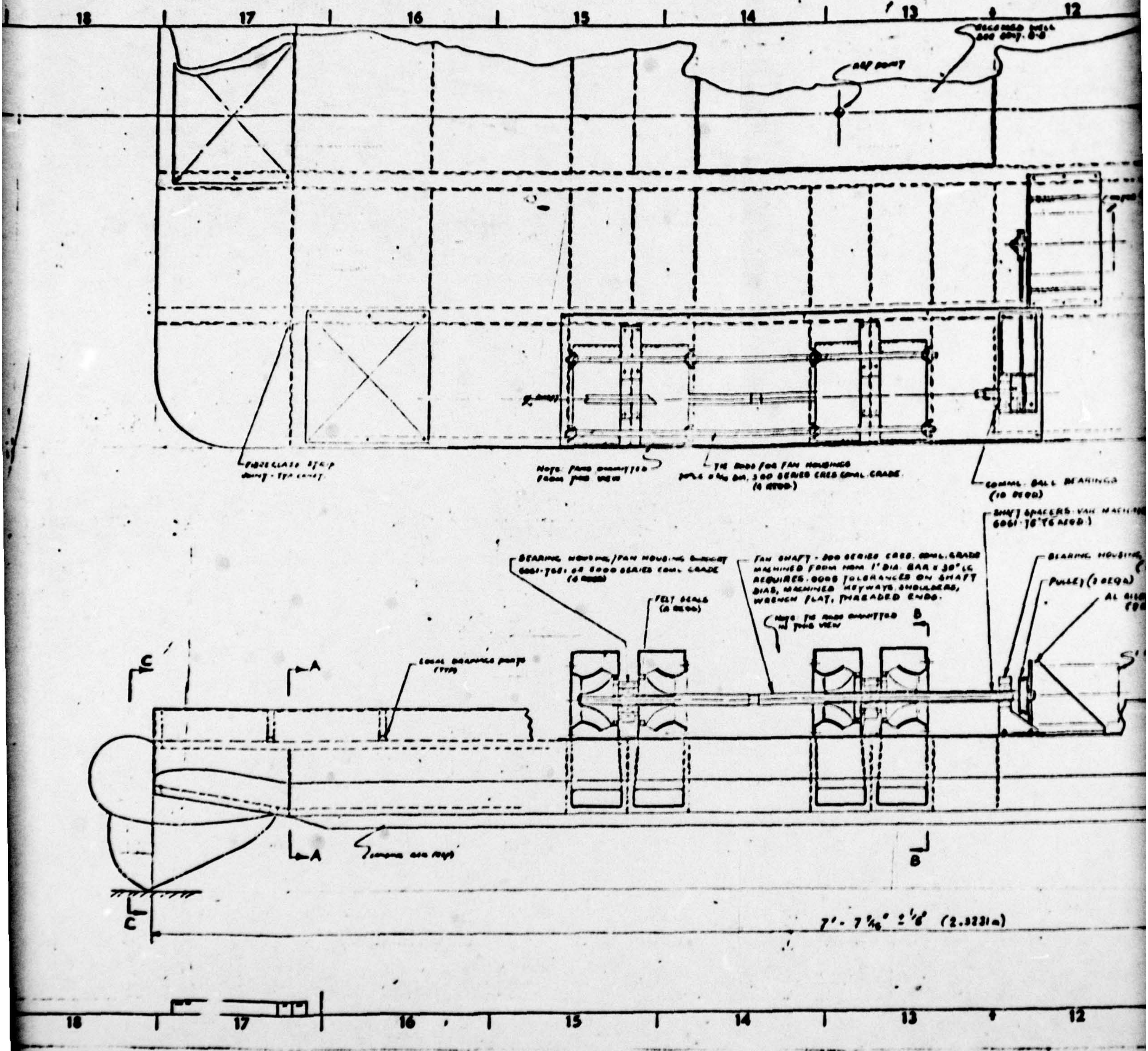
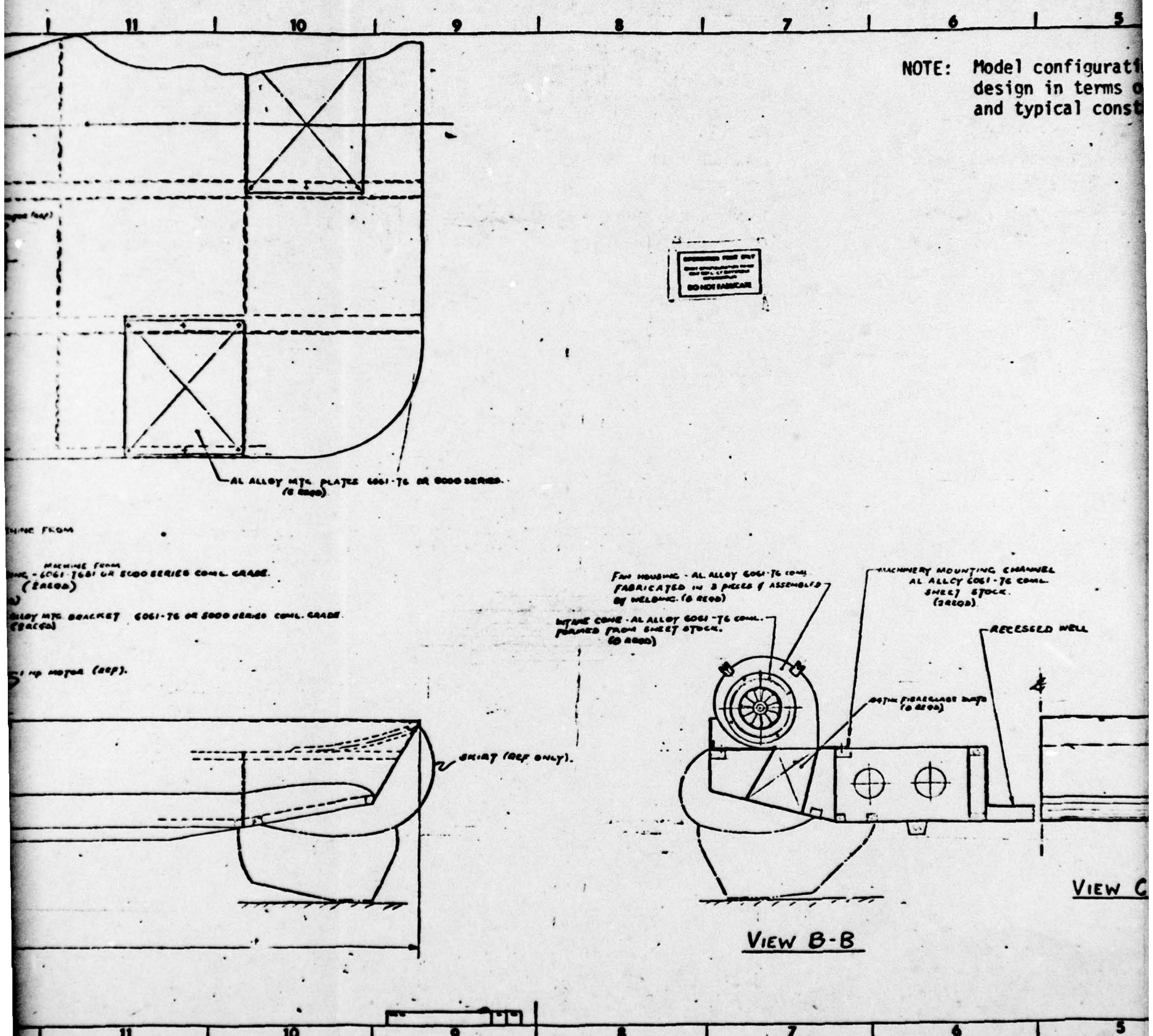
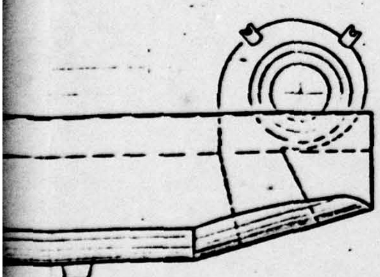


Figure 2a Cushion Comparison i



/12 Scale Model

ation shown represents suggested
s of overall size, machinery arrangement
nstruction requirements.

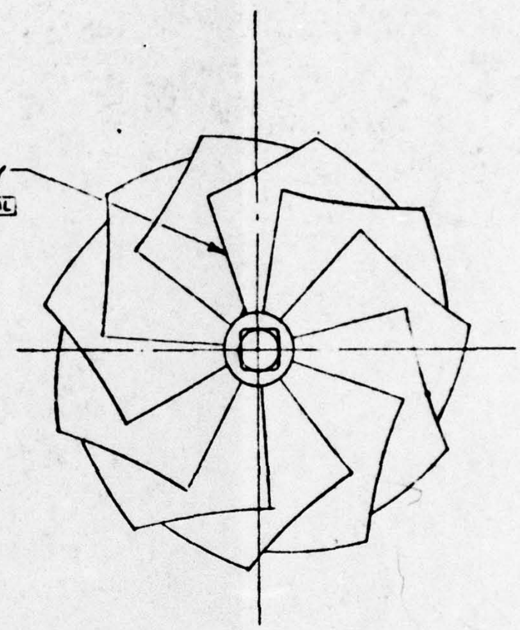


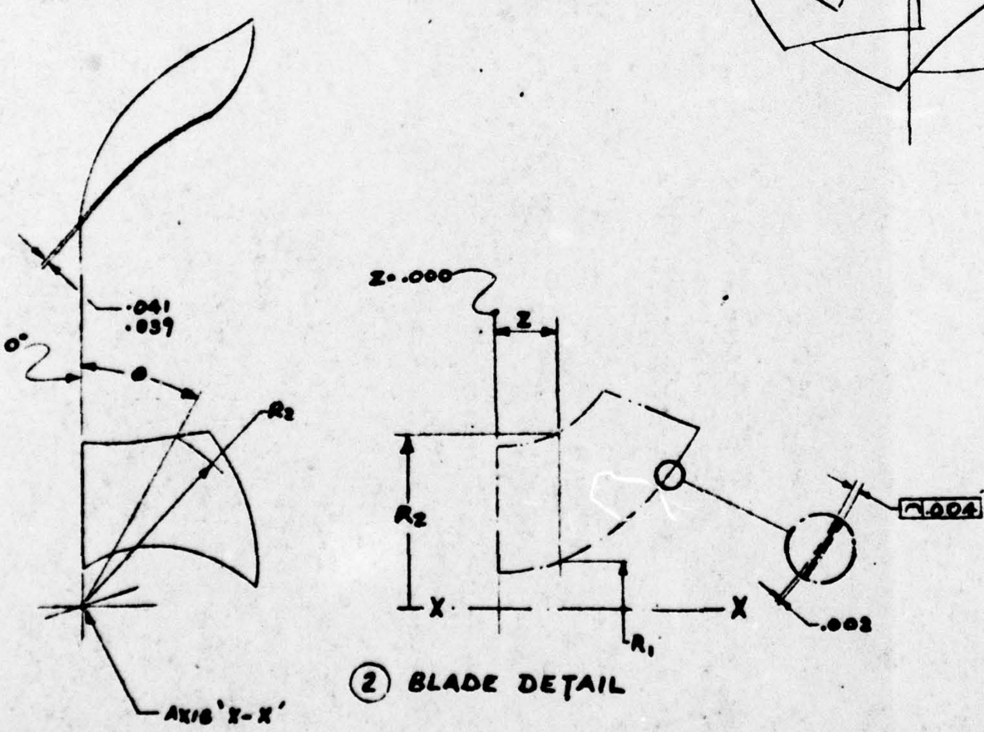
C-C (BOW SIMILAR)

CUSHION COMPARISON.
1/12 SCALE MODEL
J 05824 SK 2299 #1
1/2

NOTES.

1. ALL DIAMETERS CIRCULAR UNLESS OTHERWISE NOTED.
2. BRAZE PER MIL-B-7883, TYPE II OR IV GRADE A.
3. HEAT TREAT PER MIL-H-6088 TO T6 CONDITION AFTER BRAZING. SOLUTION HEAT TREATMENT FROM THE BRAZING OPERATION IS PERMISSABLE.
4. PENETRANT INSPECT PER ALRC-STD-4816, TYPE OPTL. NO CRACKS PERMITTED.





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9	.504
12	.680
15	.860
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24	1.364
27	1.521
30	1.673
33	1.816
36	1.909
38.42	2.063
42	2.224
45	2.356
48	2.483
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Figure 2b Model Rotor for Cushion Comparison 1/12 Scale Model

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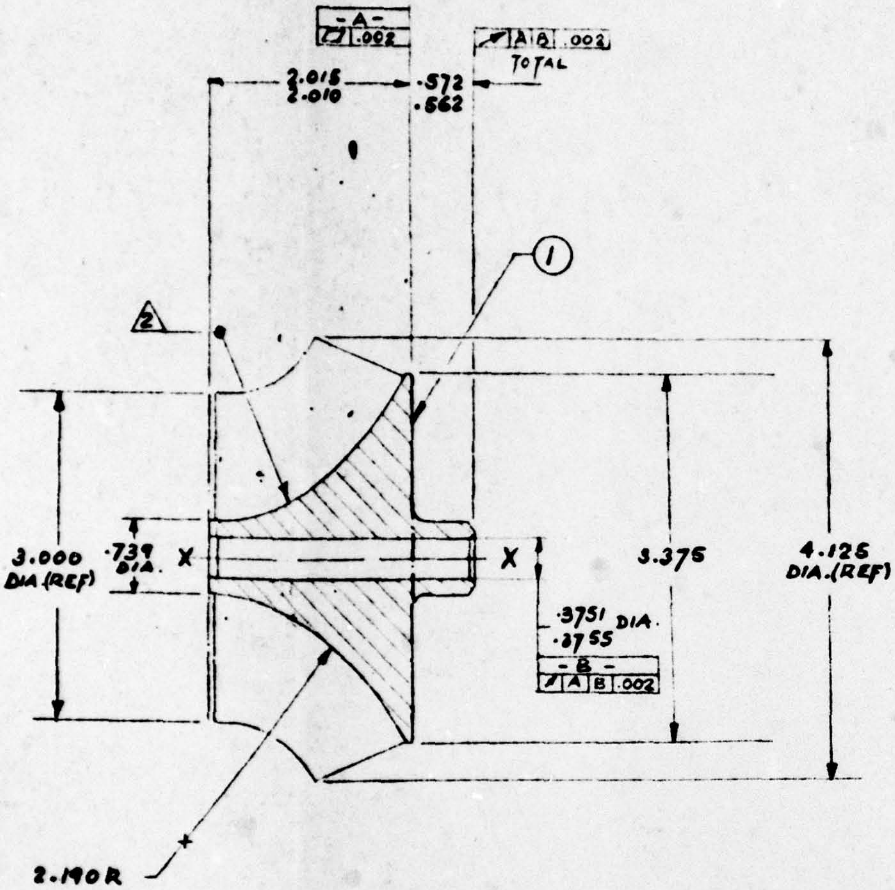

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.756	3.013
.780	3.032
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.859	3.113
.911	3.181
.969	3.265
1.030	3.370
1.106	3.492
1.179	3.637
1.256	3.790
1.336	3.960
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1.516	4.079
1.614	4.042
1.716	4.005
1.830	3.966
1.900	3.929
2.006	3.882
2.100	3.870
2.240	3.837
2.407	3.760
2.583	3.665
2.782	3.600
2.963	3.533
3.150	3.462
3.297	3.387
3.375	3.375

AR			BRAZE ALLOY	QQ-B-655 CLASS FS - BAL S.-4(4047) FORM OPTL	4
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MODEL ROTOR

SK2299 #2

2

TABLE 2

REVIEW OF SUGGESTED MODEL TEST PROGRAM

	TEST	COMMENT
1.	Model fan performance - pressure and efficiency vs flow rate at two different rpm.	One RPM at the design point is considered adequate. Effects of small change in Reynolds Number not considered significant.
2.	Model lift system performance - pressure and efficiency vs flow rate at two different rpm. Pressure losses across the system.	Same as #1
3.	Model resistance in calm water - as a function of model speed, all-up-weight, trim, heel and flow rate.	Center-of-gravity should be varied and the resulting trim and heel measured as these also vary as a function of model speed.
4.	Model stability - moment to trim and heel as a function of all-up-weight and flow rate.	Over land at zero forward speed. Over water at forward speed.
5.	Model dynamic characteristics - natural periods and damping in roll, pitch and heave as a function of all-up-weight and flow rate.	To be combined with pitch and roll stiffness tests.
6.	Model resistance and response to regular waves of varying length and height at two different flow rates.	Resistance is also dependant upon center-of-gravity which should be varied.
7.	Model resistance and response to irregular waves in Sea States 2, 3 and 4 as a function of flow rate.	Same as for #6
8.	Plow-in characteristics, trim, flow rate, wave height and frequency.	Forward speed required to characterize "plow in"
9.	Self propulsion test. Establish need for such tests.	Not considered appropriate at this time. Refer to Section 4 for details.
10.	Observation of spray pattern during the tests.	To be considered only as a model to model comparison <u>not</u> completely indicative of model to full scale.

TABLE 3

KEY PARAMETERS - FULL SCALE/MODEL

PARAMETER	FULL SCALE	1/10 SCALE	1/12 SCALE	1/14 SCALE
1. Overall length (hard structure) ft	91.46	9.15	7.62	6.53
2. Overall length (cushion inflated) ft	96	9.60	8.00	6.86
3. Overall beam (hard structure) ft	43	4.30	3.58	3.07
4. Overal beam (cushion inflated) ft	48	4.80	4.00	3.43
5. Payload deck width ft	26	2.60	2.17	1.86
6. Skirt height (mean) ft	5	0.50	0.42	0.36
7. Cargo deck height ft	5	0.50	0.42	0.36
8. All up weight (lb)				
Design	340,000	340	197	124
Maximum	390,000	390	226	142
Minimum	290,000	290	168	106
9. Mass moment of inertia - slugs - ft^2 at design all-up weight				
Roll	1.96×10^6	19.6	7.9	3.6
Pitch	6.83×10^6	68.3	27.5	12.7
Yaw	8.16×10^6	81.6	32.8	15.2
10. Transverse C.G. (from centerline) ft	0	0	0	0
11. Vertical C.G. (above baseline) ft	6.1	0.61	0.51	0.44
12. Ballast Allowance (lb)		38	22	14
13. Model bare weight (lb)		232	126	72

None of this construction is in any way representative of the full scale structure. It yields an extremely stiff model that acts essentially as a rigid body. Therefore, it cannot be used effectively to determine load conditions during wave impacts or similar data.

Further stiffness and complete watertight integrity is obtained by foam filling the entire hull volume except for the required air distribution ducts.

It was also found convenient on previous models to leave the central bay unfilled to allow space for a gimbaled pivot with its center located at deck level. This has the disadvantage that water can collect in this space and change the effective model weight. The top of the deck must be sealed with a layer of fiberglas to prevent water from soaking into the foam filling or the wooden members. Since the model is to have freeboard above the cargo deck level, additional scuppers or overboard drainage holes must be provided to allow any water to drain from the deck. Internal drains are inconvenient because of the cushion pressure acting upon the entire undersurface of the hull.

It is estimated that the weight of the hull, using the construction described above would be no more than 41 kg (90 lb). This would allow a weight of 16 kg (36 lb) for the skirts and lift fan machinery to obtain a bare model weight of 57 kg (126 lb). The desired range of running weights, center-of-gravity locations and moments-of-inertia would be obtained by the use of ballast in addition to the parts of the towing gear and instrumentation supported by the model. The ballast can be in the form of lead plates that can be rigidly attached to the model and/or standard laboratory weights. For consistency during the test program it is important that the weight and balance can be readily checked while the model is in the test rig.

The towing gear should be arranged so that the model can be lifted and supported completely clear of the water surface for weight and balance checks and to keep the skirts drained. If the model is allowed to float off-cushion for any period of time, adequate time should be allowed for any water trapped in the skirts to drain before the model is towed at high speed.

The skirts shall be of scaled geometry and represent the detail of the full scale skirt to the maximum practical extent. Particular attention should be paid to the attachments and any air leakage paths that they form.

The skirt shall be constructed of a woven fabric that is completely sealed with an appropriate coating to represent the basic characteristics of full scale skirt materials. It is recognized that structural properties cannot be modeled with true dynamic similarity but careful judgment should be exercised to make it as representative as possible. It is also recognized that drag forces generated by the skirt depend upon friction coefficients that vary with Reynolds number that cannot be maintained simultaneously with Froude number.

The lift fan performance also is dependent upon Reynolds number. If fans could be constructed with perfect geometrical similarity, their performance would not quite reach the true scaled values due to Reynolds number effects. In practice, they cannot be made with scaled accuracy and further losses are inevitable. To compensate for this, a moderate increase in speed can be utilized to obtain the required flow. However, the pressure losses throughout the system may not be quite to scale. Alternatively, the fan may be made a little larger. Since the correction cannot be predicted precisely, it will still be necessary to calibrate the system and adjust the fan speed to achieve a particular flow. Adjusting the fan scaling may also lead to compromises in other associated parts of the system such as the diffuser or ducts that distribute the lift air to the cushion.

The shafts, bearings, supports and other details of the lift fan machinery will not be to scale, but can be designed to fit within the same geometrical constraints as the full scale system.

The lift fans will be powered by one or more variable frequency motors of a nominal 1 HP, 400 Hz, 3 phase type. Since the craft configuration calls for a set of lift fans on either side it can be more convenient to provide a

motor for each side. There is greater flexibility to place the motors to obtain the most convenient balance condition. They can still be fed from a common supply or be fed and/or controlled independently. Further consideration of the test plan may lead to a requirement for independent control. If the lift fans are to be controlled independently, consideration must also be given to the use of some device to prevent return flow through idle fans.

The contemplated test plan includes self propulsion tests which would require an air propeller. A single modular unit could be placed on the deck but would require the removal of an equal weight of ballast. This may not be physically possible. An alternative would be to have a pair of propulsors that could be placed symmetrically on the deck or even have variable locations to serve the same purpose as the ballast as well as providing thrust. At this stage, it does not seem worthwhile to pursue their design, at least, until the requirements are more definitive.

5. MODEL ESTIMATED COST AND SCHEDULE

The model design and build schedule referenced in Table 4 approximates to a 4-5 month effort, past experience indicates that this is a realistic and achievable time frame. Two determining factors affecting the schedule will be the initial design effort of the model machinery package. The latter would be an 8-10 week task and would in fact be the pacing item for the completion of the model. The initial model design and drafting phase is considered to be a two man level of effort, this would be required to expedite the fabrication of the model plug and machinery package.

Obviously in terms of time and liaison effort it would be desirable to have the model built and assembled by one vendor, however, in practice this has not been possible to arrange. As the schedule indicates the model has been broken down into four components, i.e. hull - plug or pattern, hull - fiberglass, fan machinery package and skirt. Some effort has been expended in locating vendors with the necessary expertise who have already demonstrated their capabilitity.

5. MODEL ESTIMATED COST AND SCHEDULE (Cont'd)

One of the more complex and costly pieces of hardware is the machinery package. The mixed flow fan at 1/12 scale requires a specific fabrication technique to achieve the necessary accuracy and finish. The vendor located (Tecma Co.) previously manufactured a 1/6 scale mixed flow fan which necessitated considerable effort in perfecting the necessary fabrication technique.

One problem experienced has been locating the source for a 1 HP AC motor. The method that most electric motor manufacturers use is to construct a motor from standard design details to a customers requirements. An "off the shelf" light weight AC motor in this power range appears to be very difficult to locate. A cost and delivery schedule was obtained from Western Gear for a potential twelve motor order in the 1 HP range. A basic cost of $995 each was quoted (and is used in Table 5), this would only include standard production testing. The delivery schedule of 52 weeks quoted for the twelve motors would obviously not be compatible with the suggested model construction schedule. Further discussions with Western Gear resulted in the possibility of decreasing the number required to just two motors and a 3 month program. This would increase the unit cost to $1800. Further communication with the vendor regarding cost andschedule is required prior to a firm price and schedule committment. The power source for the model is considered to be the only real soft spot in the overall suggested model program.

TABLE 4

MODEL SCHEDULE

HULL

1) Model Fabrication Drawings

2) Procurement

3) Model Shell Plugs

4) Splash Mold (Fiberglass or plaster

5) Shell and Air Ducts (Fiberglass)

6) Fabricate Internal Bulkheads, Stringers

7) Fiberglass Reinforcings, Foam in Place, Machine and Add Fiberglass Top

MACHINERY

8) Fabricate Fan, Housings, Shafting, etc

9) Electric Motor-Spec/Purchase Off shelf

10) Assemble to Model, Check Out

SKIRT

11) Fabricate Skirt

12) Fit to Model, Check Out and Adjust

13) Prepare for Shipment

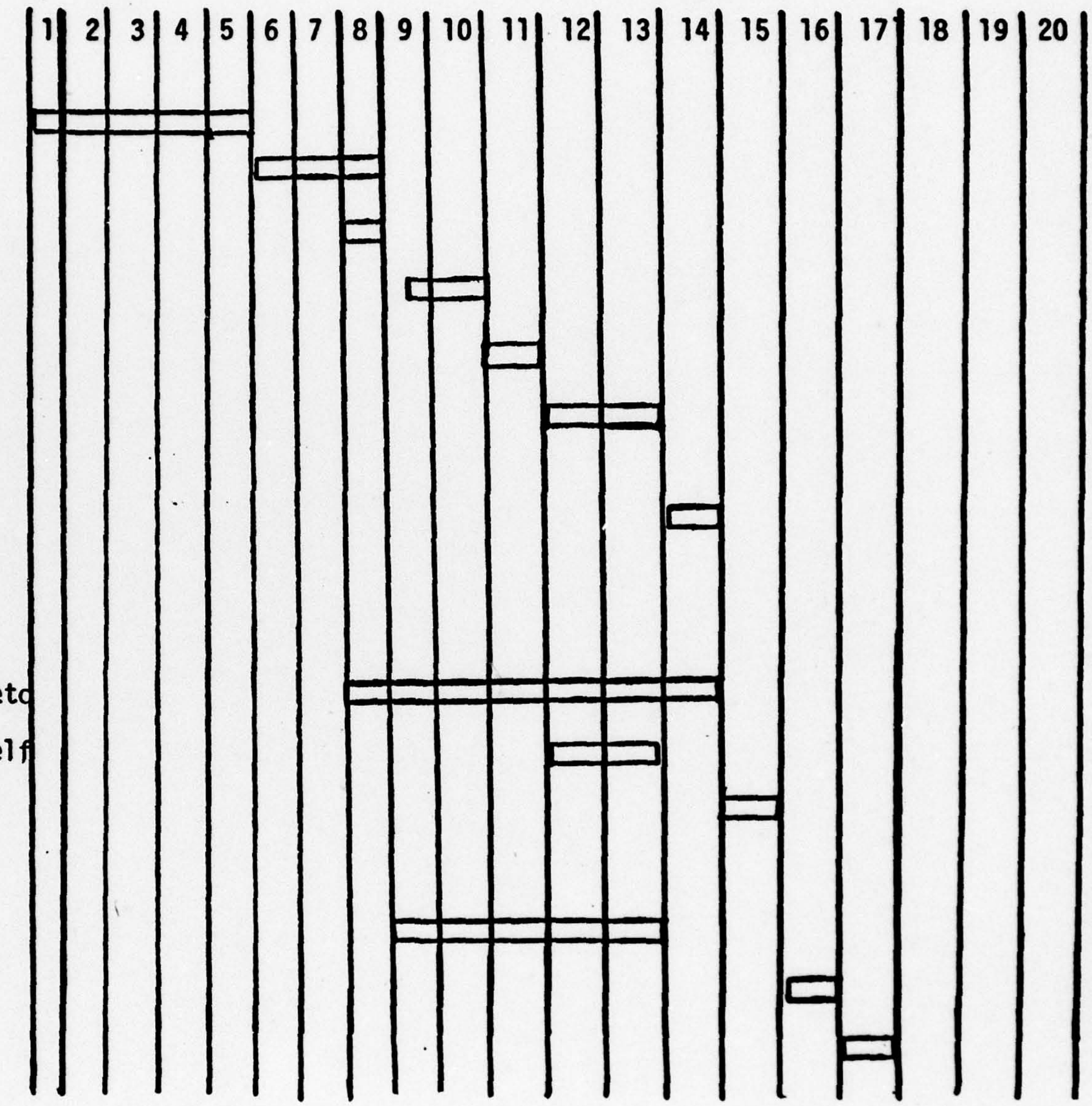

TABLE 5

ROM MODEL CONSTRUCTION & ASSEMBLY COSTS

TASK:	$ COST
1. Produce model dwgs (14 required)	12,600
2. Procurement	2,750
3. Fab model shell plug, duct plugs, internal bulkheads and stringers	7,000
4. Fab splash (fiberglas or plaster), reinforce bulkheads, foam in place, machine and add fiberglas top	5,000
5. Fabricate fans, shafts, mounting details	13,200
6. Fabricate skirt and produce layouts	7,000
7. Purchase 1 HP AC motors (2)	3,750
8. Final assembly and check out	3,500
9. Prepare for shipment	1,200
TOTAL	$ 56,000